Design and Manufacturing of steering system for ATV

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Abstract

Steering system is used to steer the vehicle according to need. In order to design a steering system it is important to recognise and compensate the forces due to tracks. Steering system of an ATV needs to be efficient as far as the parameters like rigidity, weight, spaces are concerned. As we are designing a steering system of ATV we need to consider a rough terrain. All the forces and torques encounter during the run are considered in order to design the mechanisms which will sustain these rough terrains. As the vehicle needs to be frequently steer in any direction so it is necessary that the mechanism should be responsive and also sustain the fatigue loads due to terrain. Also the steering wheel is designed with consideration on steering force. All the findings and values of the steering report is given in this paper.

Keywords**:** *Ackerman geometry, Wheelbase, Rack and Pinion.*

INTRODUCTION

Steering system is used to steer the vehicle as per need. It need to be stable, responsive, and should sustain forces on it.

As we are designing the steering mechanism of ATV we must consider factor such as weight, space, value and material selection as per its characteristics.

There are various steering available or designed in market. But as per our need we choose rack and pinion mechanism with anti -Ackerman geometry. This gives the most responsive turning of wheel with less tuning of steering.

It should be noted that the steering is an important aspect in every vehicle as it plays key role in vehicle performance, especially in racing industry.

DESIGN OF STEERING SYSTEM

The task of steering mechanism in ATV is to turn the vehicle as per drivers need and give a lower turning radius in rough terrain.

Design steps involved:

1. Identify vehicle requirement
2. Geometry set up
3. Geometry validation
4. Designing of mechanism
5. Manufacturing

Identify vehicle requirement

The initial stage of designing a steering mechanism is to know its requirement. As we are designing our vehicle for BAJA SAE event we followed some guild lines of these events. The vehicle chassis was designed according to the rule book of BAJA-2018, and following parameters were finalized based on driver safety and comfort ability:

- front track width: 56”
- 2.rear track width: 54”
- 3wheel base (l): 57”

Using this basic parameter we started our designing.

Geometry setup

It is the most important step of designing because steering geometry plays a key role in factors such as turning radius, slips, etc. also for secondary factors such as weight, cost and maintenance [1]. After a lot of research and discussion we decided to use anti-Ackerman geometry as it gives a shorter turning radius which is our main focus for the event [2].

***Table 1:** Geometry setup*

parameter	**Value**
Track width (b)	54" = 1371.6mm
wheel base (l)	56" = 1422.4mm
Steering arm length(lst)	110 mm
Ackerman angle (α)	20^0

After a discussion the geometry was finalized with the following parameters :

Table 2: Discussion the Geometry

Parameter	**Value**
Angle of inside lock (θ)	46^0
Angle of outside lock (φ)	29.5^0
Turning radius of front inner wheel (Rif)	2514.2mm
Turning radius of front outer wheel (Rof)	3885.8mm

Geometry validation

The geometry validation was done by reverse engineering and constantly validating it with calculations. Calculations performed in order to validate the geometry are as follows:

Required condition is:
Angle of inside lock (θ) = 46^0
According to Ackerman's geometry, the perfect steering condition would be:

$\cot(\varphi) - \cot(\theta) = (b \div l)$
Hence rearranging and substituting values in above equation;
$\cot(\varphi) = (b \div l) + \cot(\theta)$

$\cot(\varphi) = (1371.6 \div 1422.4) + \cot(46)$

$\boldsymbol{(\varphi) = 29.46^0}$
Also,

$(\alpha) = \tan^{-1}[(\sin(\varphi) - \sin(\theta)) \div (\cos(\varphi) + \cos(\theta))]$

$\boldsymbol{(\alpha) = 20.73^0}$
$l/\sin(\theta) = 1371.6 \div \sin(46)$
(Rif) = 2514.2 mm

(Rof) = $l \div \sin(\varphi) = 1371.6 \div \sin(29.45)$
(Rof) = 3885.8mm
(Rcg) =3200mm
As the analytical and calculated data are near about same we can say that design is accurate and will satisfy the needs of vehicle.

Design of mechanism

We used rack and pinion mechanism as it is easy to manufacture, less complex, and it also accommodate in small spaces compared to other mechanism present.
The steps involved are following:

Selection of gear tooth profile

A 20^0 full depth involute profile system was selected because it reduces the under cutting and also reduces the interfering while meshing. Due to increase in pressure angle, the tooth became broader at its base, which reduces the chances of bending failure.

The properties 20^0 full depth involute profile system are:

Table 3: *The properties 20^0*

Parameter	value
Pressure angle (φ)	20^0
Addendum (ha)	m
Dedendum (ha)	1.25m
Clearance (c)	0.25m
Working depth	2m
Whole depth	2.25m
Tooth thickness	1.5708m

Minimum number of teeth on pinion the minimum number of teeth on pinion required in order to avoid interference:

$Zp = 2*ha/ (m*\sin^2(\varphi))$

Substituting values in above equation;

$Zp = 2*m/(m*\sin^2(\varphi)) = 2/ \sin^2(\varphi)$

Zp = 17.09

Hence minimum number of teeth on pinion is 18.

Selection of material

The materials used in the steering system targets precise operation and should be light in weight. Also the secondary needs such as cost of material cost of its manufacturing and reliability is also important.

While designing the steering it is at most important that it is manufactured with high precision and given necessary tolerances to carry out smooth and effective steering of vehicle.

There are various components in steering which require different types of material for there working and needs, so steering is sub divided in following components:

1. Steering casing
2. Steering rack
3. Pinion gear
4. Steering Wheel
5. Universal Joints

The primary purpose of Steering casing is to give a support to rack and pinion and to provide mounting for the mechanism.

When the driver turns the steering wheel, the pinion gear rotates and the rack moves laterally. The rack and the knuckle and joint by an intermediate component called tie rod [3].

The tie rod ends are fitted with POS which allows the mechanism to be flexible and adjust itself with the suspension moment.

After lots of discussion we took AL6061 for casing and EN9 for the rack and pinion and, while for stub EN8 is used. All the materials selected based on the properties such as EN9 is has high surface wear resistance which is basic need of gear and rack, while AL6061 is light in weight and can sustain the force very well.

Design of gear pair

While turning steering system, mainly rack and pinion should exert required force to turn the vehicle. To overcome this rack and pinion should sustain bending and wear failure.

For this we calculated the necessary strengths.

The detailed procedure of designing is as follow:

Beam strength

The maximum tangential load a gear tooth can bear without any damage to it is known as its beam strength.

Assumptions in analysis of beam strength

Tip of single teeth bears the full load

The effect of radial force is neglected
The load is uniformly distributed over the full face width
Effect of stress concentration is neglected
Frictional force are neglected

Analytical calculations
a. bending strength of pinion (σbp): (σbp) = (Sut)/3 = 541/3 = 180
b. bending strength of gear (σbg): (σbg) = (Sut)/3 = 541/3 = 180
c. Lewis form factor (Y):

YP = 0.484 - (2.87/ Zp) = 0.484 - (2.87/17)
YP = 0.3151
Yg = 0.484 - (2.87/ Zg) = 0.484 - (2.87/29)
Yg = 0.3850
as , σbp* YP < σbg* Yg
as bending strength of gear is more than bending strength of pinion, we need to design a proper pinion first-assuming b = 10*m,
The beam strength is given by,
Pb = σbp*b*m*y
Pb = 180.33*10*m*m*0.4172
Pb = 752.336 m^2 N

Wear strength
The wear of gear tooth mostly occurs due to Pitting and Frosting. All this depends on the wear strength of the material. Hence wear resistance of the gear material should be known.

This can be calculated by Buckingham theorem.

Pw = b.Q.dp.k
Q = Rxternal gear pair factor = [(2.Zg) / (Zg + Zp)]
Here Zg = 28 , Zp = 18 So that Q = 1.20
k = [σc^2. Cosϕ Sinϕ. (1/ϵ1 + 1/ϵ2)] / 1.4
k= { [(0.27). (9.81).(BHN) $]^2$. Cos(20).Sin(20). (1/71700)}/1.4
k = 0.45. (BHN/100$)^2$
k= 0.45. (150/100$)^2$
k = 1.0125
Hence wear strength can be calculated as follow;
Pw = 255.15m^2 N

Effective load
Effective load was calculated based on requirement of the system. While cornering the drive applies an effort to steer the vehicle , this force be more to overcome the frictional forces generated between the road and tyres.
As human tents to apply brakes while turning, we considered 50% distribution of weight.
Thus the reaction forces are computed based on above assumptions are calculated as such-
So, FT = 50% weight of vehicle = 833.85 N
on both suspension joints.

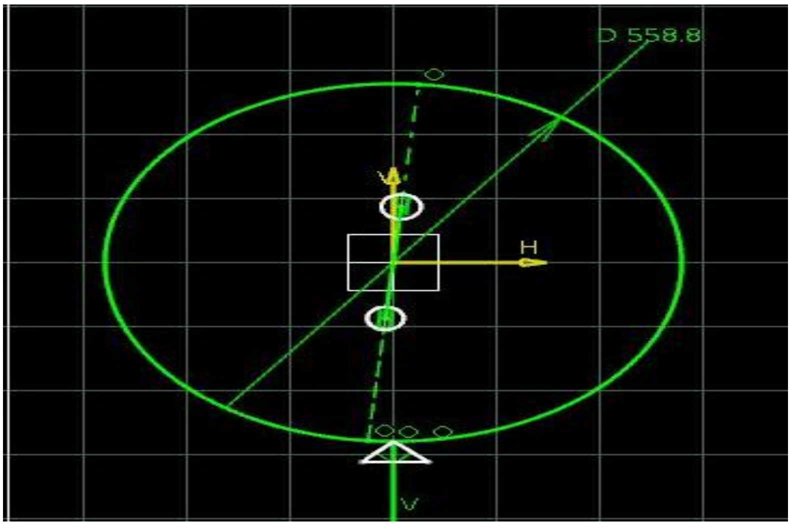


Figure 1: *castor trail and scrub radius calculations*

From this we can also deduce that,
FZR = 0.5 (FT)
FZR = 416.925N
Required torque to overcome the steering axis inclination is given by;

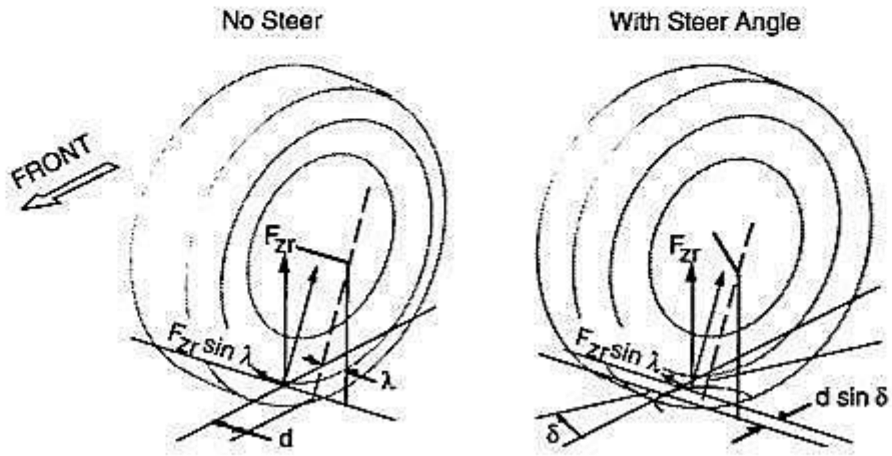


Figure 2: *Lateral forces on wheel while*

cornering Moment to overcome offset of SAI axis-
=(FZR + FZL). d.Sinλ.Sinδ
=(833.85).70.Sin4.Sin46
=-2928.90 N-mm
Aligning torque required to compensate caster trail offset is given by-

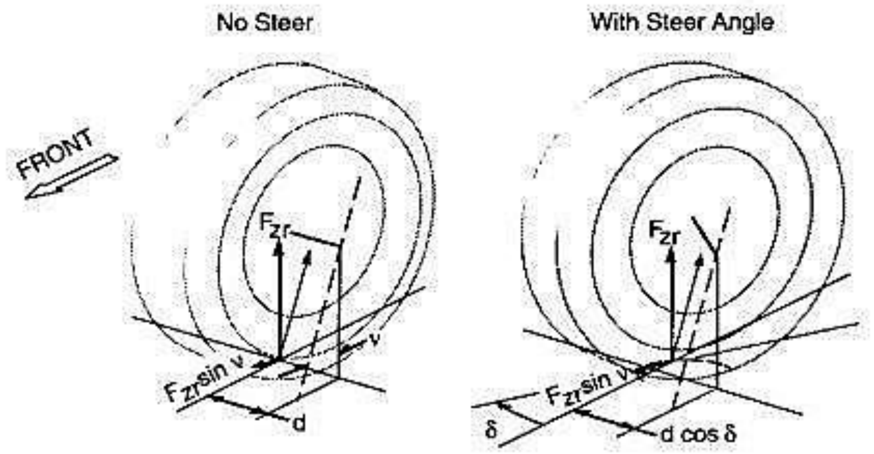


Figure 3: *caster trail*

Moment to overcome the offset of caster trail
=(FZR – FZL)*d*Sinμ*Cosδ
= (416.925)*70*Sin5*Cos46
=1766.945 N-mm
hence,
MT = -2928.90-(1766.945)

MT = 4695.845 N-mm

The friction couple generated while cornering -
friction couple per wheel = m*v^2/ 4*Rcg
=[170*((5/18)*v)^2]/ 4*2.34
= 1317.739 N

Therefore,
Torque for friction couple (Mf) is given by-

Torque (Mf) = 1317.739*(R/2)* Sinμ.* sinδ
=1317.739*(279)*sin(5)*sin(46)
=23053.795 N-mm

Meff = (Mf + MT)
=(23053.795+4695.845)
=27749.64 N-mm

The tangential force across the pinion is-
Meff = Ft * steering arm length
Hence; **Ft = 252.26 N**
From design data book;
For accurate mounting and moderate shocks :
Ka = Application factor = 1
Km = Load concentration factor = 1.3
Kv = velocity factor = 1
Peff = (Ka * Km)* **Ft** / Kv
Peff = 327.935 N

Estimation of module
As gear pair is weaker in wear than in bending so the parts must be designed considering wear failure,

Assuming factor of safety (FOS) (Nf) = 1.8 For given system ;
Pw = Nf * Peff
255.15m^2 = 1.8*327.925

m = 1.5 mm
The module is estimated by the basic parameters of gear design is given by;

Dimensions of pinion

Table 4: *Dimensions of pinion*

Parameter	Pinion
d (mm)	42.5
Addendum(ha)	1.5
Dedendum (hf)	1.875
Face width(b)	25
Circular pitch(Pc)	7.85
Module(m)	2.5
Zp	18

Dimensions of Rack
Zg=28
F. Virtual prototyping of system using CAD software
We did virtual prototyping using CATIA V5R20.
As per the dimension obtain by calculations, 3D models of pinion and rack were generated in CATIA.
dimensions, weight and assembly of the system.
The 3D CAD models are shown

CAD model of pinion

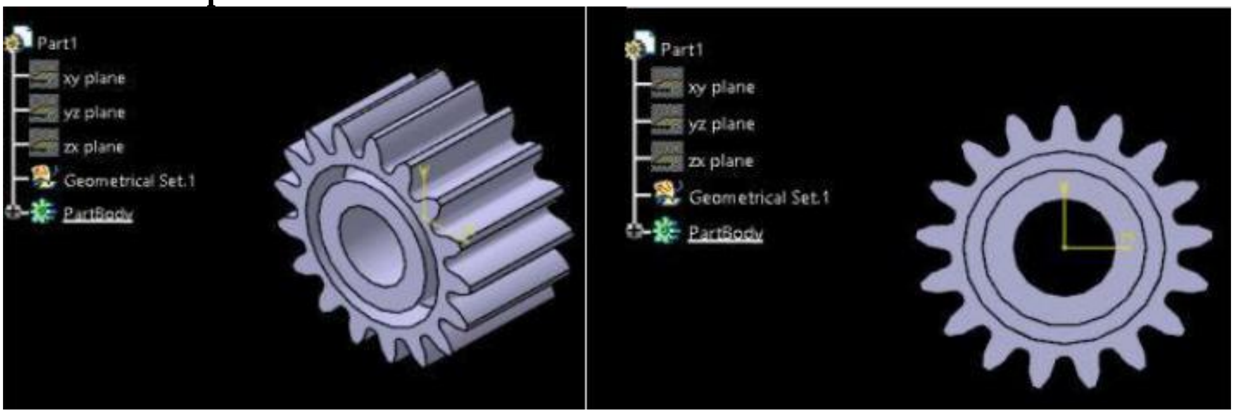


Figure 4: *Cad model of pinion*

CAD model of rack

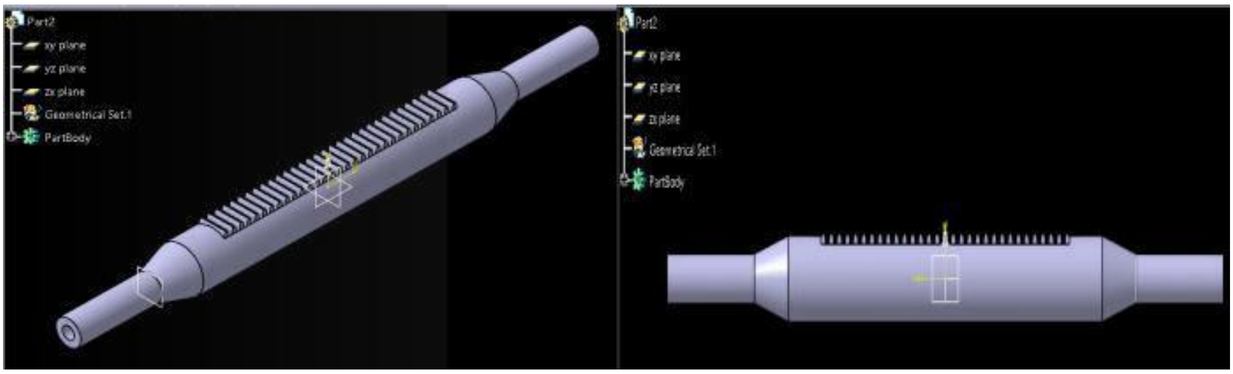


Figure 5: *Cad model of rack*

CAD model of rack and pinion assembly

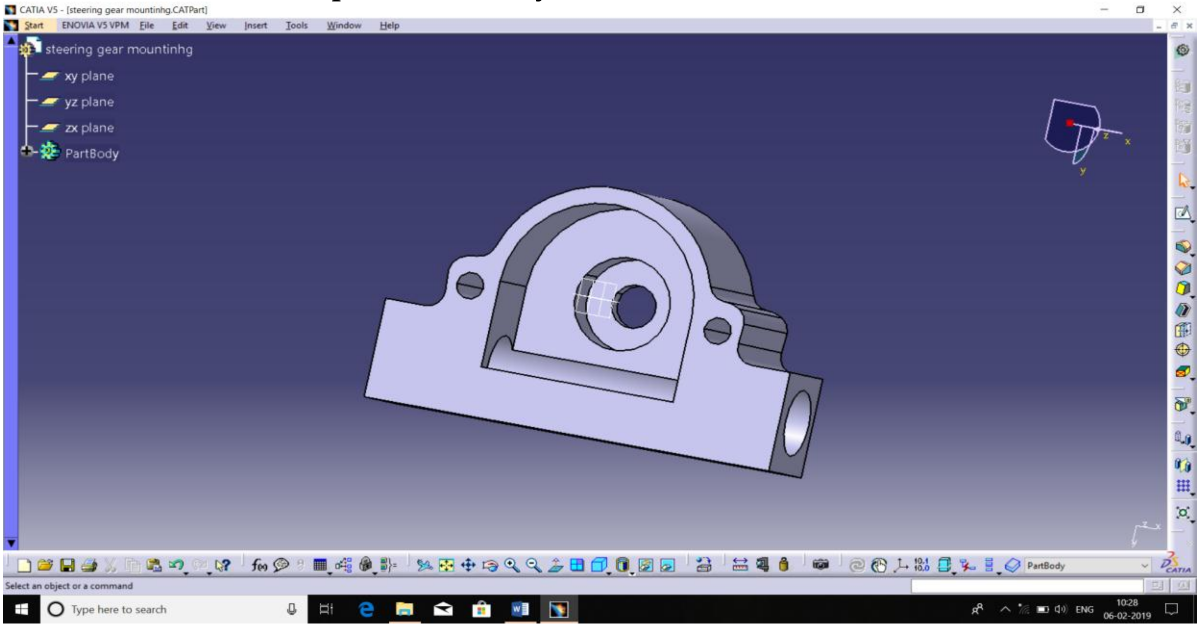


Figure 6: *Steering casing*

CAD model of steering system

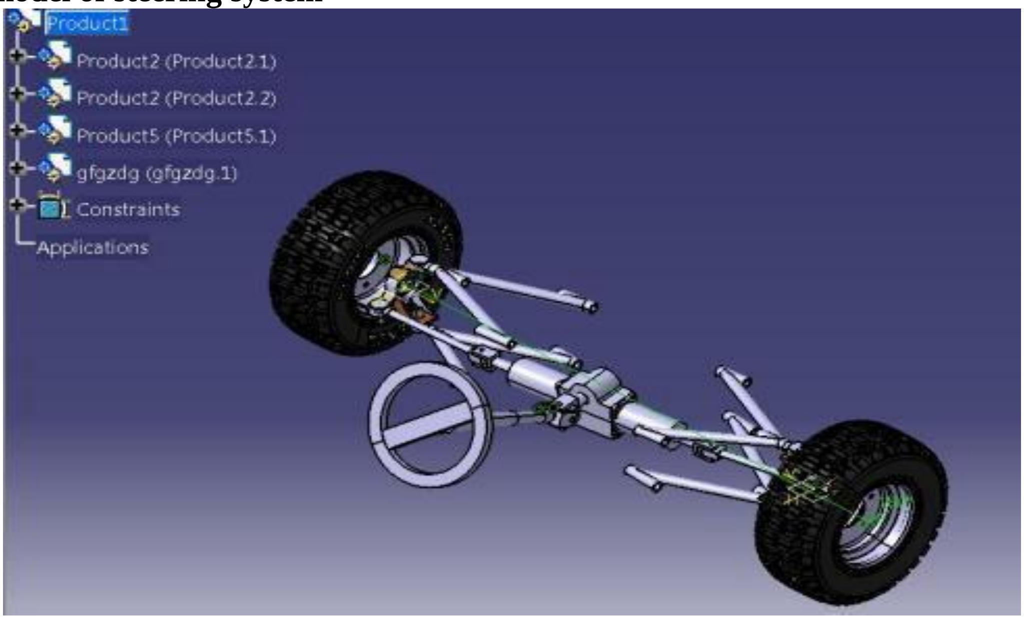


Figure 7: *Assembly of steering system*

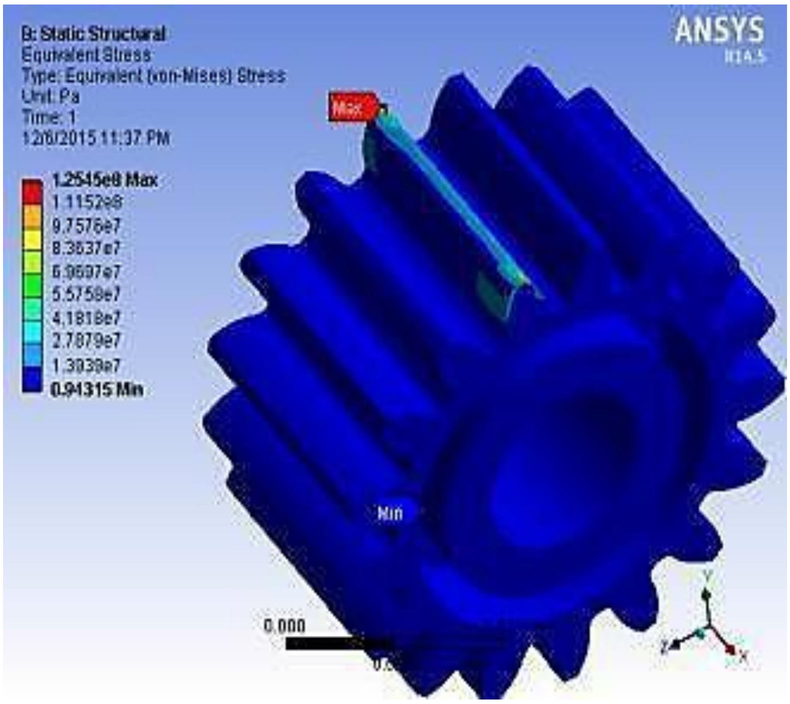
FEA of system using ANSYS software

ANSYS was used to validate the desire result and based on solution; we did necessary changes in order to improve strength hence, reliability of the system.

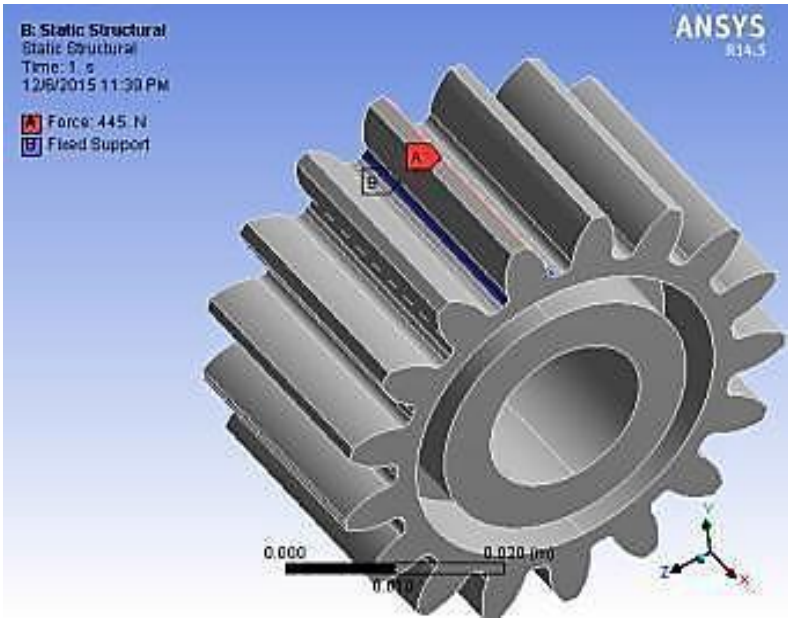
The Finite Element Analysis was performed using CATIA.

The results of analysis are tabulated below:

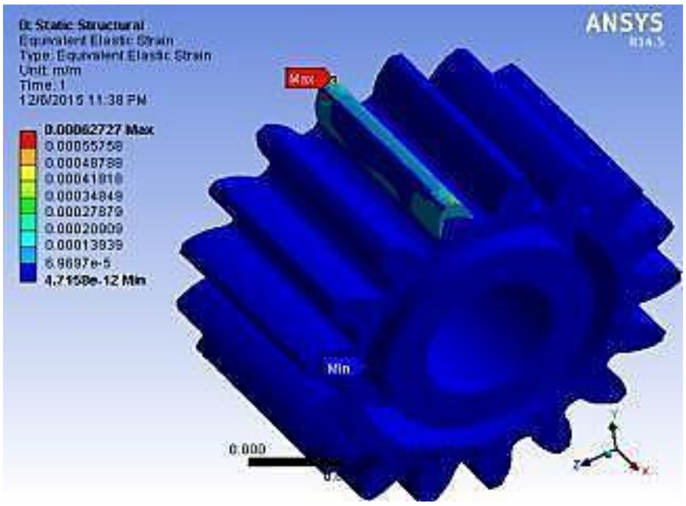
FEA of pinion



***(a)** Mesh modelling*

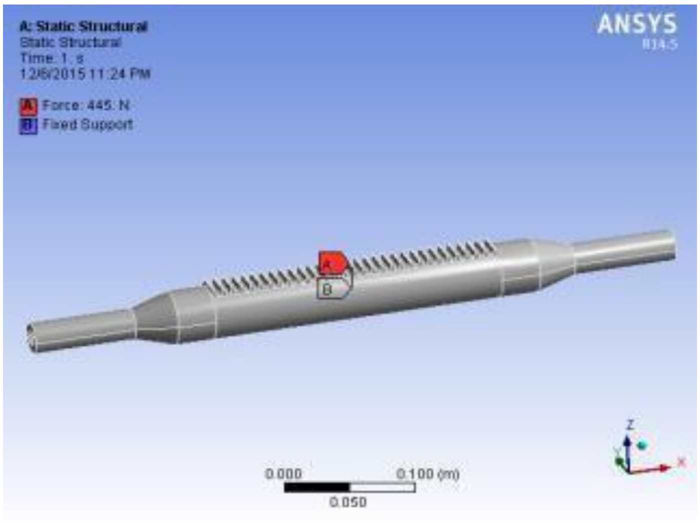


***(b)** Stress*

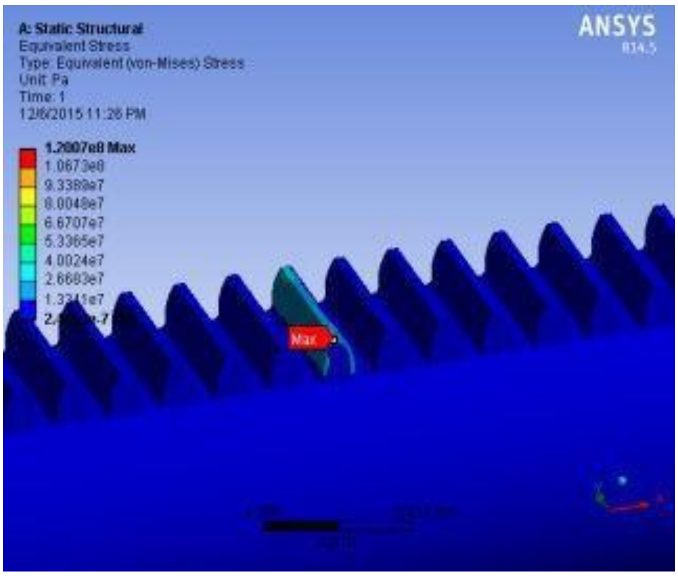


(c): Deformation:

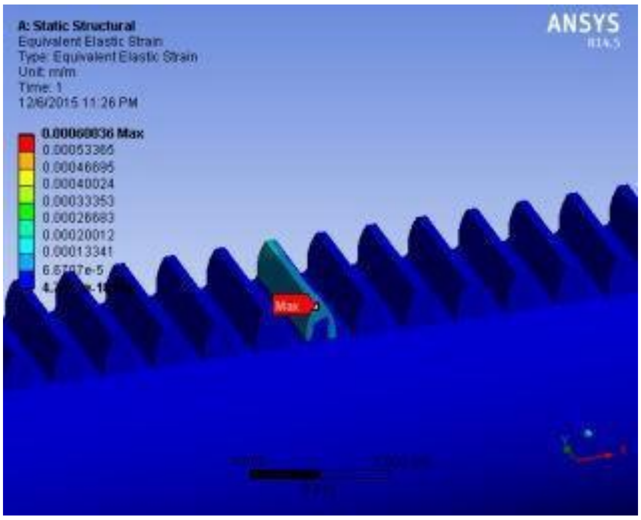
FEA of Rack



(d) Mesh modelling



(e): *Stress*



(f) ***Figure 8:*** *Deformation*

Table 5: *Result of ANSYS*

Parameter	**Value**
Stress	120.07MPa
Deformation	0.600mm
FOS	3.33
Parameter	**Value**
Stress	125.45 Mpa
Deformation	0.6277 mm
F.O.S	3.18

(a): ***(b)*** ***Figure 9:*** *Manufacturing and testing*

CONCLUSION

Hence we conclude following points

1. This project work on the modification on the previous steering designed used in our last year car.
2. We came to know that smallest change in the design brings a drastic change in steering of vehicle.
3. Hence we improved the performance and reduced the weight of the component without affecting its strength.

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